VOLUME CONTROL APPARATUS OF RADIAL PISTON PUMP OR MOTOR AND POSITIONING APPARATUS

FIELD OF THE INVENTION

The present invention relates to a volume control apparatus of a radial piston pump or a motor in which each of pistons is arranged so as to slide in a radial direction with respect to a rotation axis, and a positioning apparatus for a cam ring of the radial piston pump or a swash plate of an axial piston pump.

BACKGROUND OF THE INVENTION

In a hydraulic working machine such as a construction machine or the like, there is mounted a hydraulic pump and a hydraulic motor, in order to drive an upper revolving body, a lower traveling body and the like.

As one kind of the hydraulic pump, there is a radial piston pump in which each of pistons is arranged so as to slide in a radial direction with respect to a rotation axis. In the radial piston pump, a cam ring is pressed by an actuator for regulating a volume, and a center of the cam ring is positioned at a position which is eccentric with respect to a center of the rotation axis (or a main axis). A volume (cc/rev) is determined in correspondence to an eccentric amount of the cam ring.

Further, as one kind of the hydraulic pump, there is an axial piston pump in which each of the pistons is arranged so

as to slide in parallel to the rotation axis. In the axial piston pump, a swash plate is oscillated by an actuator for regulating a volume, and the swash plate is positioned at a tilted position with respect to the rotation axis (or a main axis). A volume is determined in correspondence to a tilting amount of the swash plate.

In this case, in recent years, in the case that the hydraulic pump or the like is mounted on the construction machine or the like, there is a request of making a weight of the hydraulic pump light by making a place product of the hydraulic pump itself small on the basis of a constraint of a mounting space, a demand from a market and the like, and improving a freedom for arranging the hydraulic pump. Accordingly, a downsizing and a weight reduction are required in the volume regulating actuator mounted to the hydraulic pump. The same matter is applied to the hydraulic motor.

The present invention is made by taking the actual condition mentioned above into consideration, and a first achieving object of the present invention is to downsize a radial piston pump or a motor, reduce a weight and improve a freedom in arrangement.

Further, a second achieving object of the present invention is to reduce a weight by downsizing a positioning apparatus for positioning a cam ring of a radial piston pump or a motor, and a swash plate of an axial piston pump or a motor.

A general technical level in connection with the

achieving object of the present invention is as follows.

In a document listed up as one example of a prior art: Japanese Unexamined Patent Publication No. 11-50968, there is disclosed a positioning apparatus in which a position of a cam ring in a radial piston pump is detected by a distance sensor, a detection signal of the distance sensor is amplified by an amplifier, the amplified signal is incorporated as a feedback amount into a servo valve, and the cam ring is positioned at a target position by controlling so as to drive the piston by the servo valve.

Since the positioning apparatus described in the publication is constituted by the distance sensor, the amplifier, the servo valve and the piston, there is a problem that a place product is increased in the case of being used as the actuator for regulating the volume of the radial piston pump.

SUMMARY OF THE INVENTION

In order to achieve the first achieving object, in accordance with a first aspect of the present invention, there is provided a volume control apparatus of a radial piston pump or a motor for regulating a volume by positioning a cam ring of the radial piston pump or the motor, comprises a control valve positioned at a position in correspondence to a volume control pressure, and a servo piston having said control valve built-in, being operated following to the control valve and pressing the cam ring so as to position the

cam ring.

In the volume control apparatus in accordance with the first aspect, as shown in Fig. 1, a servo piston 8 is operated following to a control valve (a spool) 9, presses a cam ring 2 so as to position the cam ring 2 at a position in correspondence to a volume control pressure, and regulates a volume. Accordingly, the same servo mechanism as that of the prior art can be achieved. Further, since the volume control apparatus has the control valve 9 built in the servo piston 8, the place product becomes small, and the weight becomes light. Therefore, the radial piston pump or the motor is downsized, the weight is reduced, and a freedom in arrangement is improved.

In accordance with a second aspect of the present invention, there is provided a volume control apparatus of a radial piston pump or a motor as recited in the first aspect, wherein one set of the control valve and the servo piston and another set of the control valve and the servo piston are provided at opposing positions with respect to the cam ring.

In accordance with the second aspect, as shown in Fig. 5A, the control valve (a spool) 9 and the servo piston 8, and a control valve (a spool) 19 and a servo piston 18 are provided at opposing positions with respect to the cam ring 2, and the cam ring 2 can be made eccentric in both sides with respect to a center of a piston valve 5. Accordingly, as shown in Fig. 5B, in the case that the volume control apparatus is applied to an alternating type hydraulic pump 61

which can change a discharging direction to two directions, it is possible to regulate the volume in both discharging directions on the basis of a small place product.

In order to achieve the second achieving object, in accordance with a third invention, there is provided a positioning apparatus comprises a control valve positioned at a position in correspondence to a volume control pressure, and a servo piston being said control valve built-in, being operated following to the control valve and pressing a positioning member so as to position the positioning member.

In order to achieve the second achieving object, in accordance with a fourth invention, there is provided a positioning apparatus comprises a control valve carrying out a stroke in correspondence to a control pressure applied to a pressure receiving surface, and a servo piston having the control valve built-in and pressing a positioning member in correspondence to a driving pressure,

wherein a throttle is formed between the control valve and the servo piston, in such a manner that the driving pressure introduced to the servo piston is increased in accordance that the control valve carries out the stroke relatively close to the positioning member with respect to the servo piston, and the driving pressure introduced to the servo piston is reduced in accordance that the servo piston carries out the stroke relatively close to the positioning member with respect to the control valve, and

wherein a spring for generating a spring force opposing

to the control pressure is applied to the control valve, the control pressure is applied to the pressure receiving surface so as to carry out a stroke of the control valve, the servo piston carries out a stroke following to the control valve on the basis of the driving pressure introduced via the throttle, the control valve is positioned at a position where the spring force of the spring and the control pressure are balanced, and the servo piston is positioned in accordance the positioning of the control valve.

In the positioning apparatuses in accordance with the third aspect and the fourth aspect, as exemplified in Figs. 1 and 6, the servo piston is operated following to the control valve (the spool) 9, presses a positioning member (a cam ring or a swash plate) 2 or 50, and positions the positioning member 2 or 50 at a position in correspondence to the control pressure. Since the positioning apparatus has the control valve 9 built in the servo piston 8, the place product becomes small and the weight becomes light. Accordingly, the radial pump or the motor, and the axial piston pump or the motor are downsized, the weight thereof is reduced, and a freedom in arrangement is improved.

BRIEF DESCRIPTION OF THE DRAWINGS

Fig. 1 is a view showing an entire structure of a radial piston pump in accordance with an embodiment of the present invention;

Fig. 2 is a view showing a detailed structure of a

volume control apparatus of the radial piston pump shown in Fig. 1;

Fig. 3 is a view showing details of a volume control apparatus having a different structure from that shown in Fig. 2;

Fig. 4 is a view showing details of a volume control apparatus having a different structure from those shown in Figs. 2 and 3;

Figs. 5A and 5B are views showing a structure embodiment in which volumes in both discharging directions of a radial piston pump are changed; and

Fig. 6 is a view showing a structure embodiment in the case that a volume control apparatus in accordance with the present embodiment is applied to an axial piston pump.

BEST MODE FOR CARRYING OUT THE INVENTION

A description will be given below of an embodiment of a volume control apparatus and a positioning apparatus in accordance with the present invention with reference to the accompanying drawings.

Fig. 1 shows an entire structure of a radial piston pump in accordance with the present embodiment. The radial piston pump in Fig. 1 is used as a driving pressure source for a hydraulic motor mounted on a construction machine and driving an upper revolving body, or a hydraulic motor driving a lower traveling body.

Fig. 1 is a view of a radial piston pump 1 as seen

along a vertical cross section to a rotation axis, and shows an eccentric type radial piston pump. The radial piston pump 1 is structured such as to receive a cylinder block 3 and the like in an inner portion of a case 14.

The cylinder block 3 is integrally formed with a rotation axis (not shown). A cylindrical piston valve 5 is fitted and fixed to the case 14 in accordance with an arrangement aspect that a rotation axis and a center axis are set identical.

A pump port P is formed in the piston valve 5 over a predetermined circumferential length along a circumferential direction of the piston valve. The pump port P is open to an outer peripheral surface of the piston valve 5. Further, a suction port S is formed in the piston valve 5 over a predetermined circumferential length along a circumferential direction of the piston valve. The suction port S is open to the outer peripheral surface of the piston valve 5.

A plurality of bores are formed in the cylinder block 3 at an even pitch in a radial direction of a rotation axis. A piston 4 is slidably provided within each of the bores. A shoe 49 is slidably connected to each of the pistons 4. A cam ring 2 is arranged in an outer side of the shoe 49. The cam ring 2 is arranged such that an inner peripheral surface can slide against a sliding surface of each of the shoe 49.

The case 14 is provided with a servo piston 8 and an opposing piston 7 constituting a volume control apparatus, in such a manner as to hold a rotation axis therebetween. The

servo piston 8 and the opposing piston 7 press the cam ring 2 so as to support in such a manner as to freely eccentrically move a center of the cam ring 2 with respect to a center of the rotation axis. A bearing 6 for slidably moving the cam ring 2 is arranged between the cam ring 2 and the case 14.

Further, a cylinder side port 4a communicating with each of the bores is formed in the cylinder block 3. The cylinder side port 4a is open to positions opposing to a pump port P and a suction port S in the side of the piston valve 5.

When the rotation axis is driven so as to rotate by a driving source, for example, an engine, the cylinder block 3 is relatively rotated with respect to the piston valve 5. Accordingly, the shoe 49 slides along an inner periphery of the cam ring 2.

The servo piston 8 and the opposing piston 7 is operated, whereby the center of the cam ring 2 is made eccentric with respect to the center of the rotation axis at a predetermined eccentricity amount. Accordingly, when the piston 4 is positioned at a position where the piston valve 5 and the cam ring 2 are closest, the piston 4 is in a top dead center state. When the piston 4 is rotated further half along the circumferential direction of the piston valve 5 from the position, the piston 4 is in a bottom dead center state at a position where the piston valve 5 and the cam ring 2 are most apart from each other. When the piston is rotated further half, the piston 4 becomes in the top dead center state from the bottom dead center state. As mentioned above,

the piston 4 carries out one stroke (the top dead center-the bottom dead center-the top dead center) every one rotation along the circumferential direction of the piston valve 5, and an amount of one stroke corresponds to twice an amount of eccentricity. In the course of one stroke of the piston 4, a pressure oil at a volume (cc/rev) corresponding to the stroke amount is sucked and then discharged.

In other words, when the piston 4 is positioned at the position where the cylinder side port 4a is communicated with the suction port S, the pressure oil is sucked from a tank into the bore via the suction port S and the cylinder side port 4a. Next, when the piston 4 is positioned at the position where the cylinder side port 4a is communicated with the pump port P, the pressure oil compressed by the piston 4 is discharged from the bore via the cylinder side port 4a and the pump port P, and is supplied to a hydraulic actuator in an outer portion. As mentioned above, the pressure oil at the volume corresponding to the amount of eccentricity of the cam ring 2 is supplied to the hydraulic actuator in the outer portion via the pump port P.

The opposing piston 7 is slidably provided in the case 14, an oil chamber 28 is formed in an inner side of the opposing piston 7, and a spring 27 is attached. In the opposing piston 7, a thrust is generated in correspondence to a hydraulic pressure within the oil chamber 28 and a spring force of the spring 27, and the cam ring 2 is pressed against the side of the servo piston 8.

The servo piston 8 is slidably provided in the case 14, and an oil chamber 20 is formed in an inner side of the servo piston 8. In the servo piston 8, a thrust is generated in correspondence to a hydraulic pressure within the oil chamber 20, and the cam ring 2 is pressed against the side of the opposing piston 7. In this case, a fixed driving pressure is supplied to an inner side of the oil chamber 28 in the side of the opposing piston 7, and a fixed thrust is generated in the opposing piston 7.

On the contrary, the driving pressure within the oil chamber 20 in the side of the servo piston 8 is changed in accordance with a volume control pressure supplied to the pilot port 12, and the thrust of the servo piston 8 is changed in accordance with the volume control pressure. Accordingly, the cam ring 2 is made eccentric at the position in correspondence to the volume control pressure supplied to the pilot port 12 of the servo piston 8. In accordance that the cam ring 2 is moved to the opposing piston 7 by the servo piston 8, the volume is reduced from the maximum volume.

Fig. 2 is an enlarged view of the servo piston 8 in Fig. 1, and shows a volume control apparatus in accordance with the present embodiment.

A spool 9 corresponding to the control valve is built in the servo piston 8 in such a manner as to be slidable with respect to the servo piston 8. The pilot port 12 to which the pilot pressure serving as the volume control pressure is supplied is formed on an outer peripheral surface of the

servo piston 8. In the servo piston 8, there is formed a pilot pressure introducing oil passage 21 for introducing the pilot pressure supplied to the pilot port 12 to an inner side of the servo piston 8.

Further, an original pressure port 13 to which the driving pressure for driving the servo piston 8 is supplied is formed on an outer peripheral surface of the servo piston 8. In the servo piston 8, there is formed a driving pressure introducing oil passage 22 for introducing the driving pressure supplied to the original pressure port 13 to an inner side of the servo piston 8. Further, in the servo piston 8, there is formed a tank discharging oil passage 24 communicating a tank 26 with the inner side of the servo piston 8.

The spool 9 is provided with a small diameter portion having a diameter D1 and a large diameter portion having a diameter D2, and is provided with a pressure receiving surface 9a formed as a step between the small diameter portion and the large diameter portion. The pressure receiving surface 9a has a pressure receiving area corresponding to a difference of pressure receiving areas $((D2) - (D1))\pi/4$ between the small diameter portion and the large diameter portion.

The pressure receiving surface 9a of the spool 9 is formed at a position in correspondence to the pilot pressure introducing oil passage 21. Accordingly, the pilot pressure is applied to the pressure receiving surface 9a of the spool

9 from the pilot port 12 via the pilot pressure introducing oil passage 21. The spool 9 carries out a stroke toward the cam ring 2 in accordance with the pilot pressure applied to the pressure receiving surface 9a thereof.

A spring 11 and a spring 10 which are expanded and contracted in the same direction as the stroke direction of the spool 9 are received in an inner side of the spool 9.

One end of the spring 11 is brought into contact with the servo piston 8, and another end of the spring 11 is brought into contact with the spool 9. Further, one end of the spring 10 is brought into contact with the spool 9, and another end of the spring 10 is brought into contact with an adjusting screw 15. The adjusting screw 15 is fixed to the case 14 via a lock nut 16.

A spring chamber in which the spring 11 is received is defined by the servo piston 8 and the spool 9, and forms the oil chamber 20. An oil passage 9d for communicating the oil chamber 20 (the spring chamber of the spring 11) with a spring chamber in which the spring 10 is received is formed in an inner side of the spool 9.

An oil passage 9c for communicating the inner oil chamber 20 with an outer side of the spool 9 is formed in the spool 9. The oil passage 9c is formed at a position corresponding to the tank discharging oil passage 24. A throttle 25 is formed between the oil passage 9c and the tank discharging oil passage 24. When the pressure of the pilot port 12 is reduced, the spool 9 carries out a stroke toward

an upper side in the drawing, that is, an opposite side to the cam ring 2. An opening area of the throttle 25 is increased in accordance with a stroke of the spool 9 toward an opposite side to the cam ring 2, and the pressure oil is discharged from the oil chamber 20 to the tank 26 via the oil passage 9c, the throttle 25 and the tank discharging oil passage 24. Accordingly, since the driving pressure within the oil chamber 20 is reduced and the thrust of the servo piston 8 is reduced, the cam ring 2 carries out a stroke toward an upper side on the basis of the thrust of the opposing piston 7. Therefore, the servo piston 8 carries out a stroke toward an upper side. The opening area of the throttle 25 is reduced in accordance with a stroke of the servo piston 8 toward an upper side, and the reduction in the driving pressure within the oil chamber 20 is inhibited. Accordingly, the servo piston 8 carries out a stroke toward an upper side in the drawing at a moving amount of the spool 9.

An oil passage 9b for communicating the spring chamber of the inner spring 10 with the outer side of the spring 9 is formed in the spool 9. The oil passage 9b is formed at a position corresponding to the driving pressure introducing oil passage 22. A throttle 23 is formed between the oil passage 9b and the driving pressure introducing oil passage 22.

When the pressure of the pilot port 12 is increased, the spool 9 carries out a stroke toward a lower side in the

drawing, that is, toward the cam ring 2. An opening area of the throttle 23 is increased in accordance with the stroke of the spool 9 toward the cam ring 2, and the pressure oil supplied from the original pressure port 13 to the oil chamber 20 via the driving pressure introducing oil passage 22, the throttle 23, the oil passage 9b, the spring chamber of the spring 10 and the oil passage 9d is increased.

A thrust in correspondence to the driving pressure within the oil chamber 20 is generated in the servo piston 8, and presses the cam ring 2. Since the pressure receiving area of the opposing piston 7 is smaller than the pressure receiving area of the servo piston 8, the cam ring 2 carries out a stroke toward a lower side in the drawing on the basis of the thrust generated in the servo piston 8, and the servo piston 8 also carries out a stroke toward a lower side. When the servo piston 8 carries out the stroke toward the lower side, the opening area of the throttle 23 is reduced, and the increase in the driving pressure within the oil chamber 20 is inhibited. Accordingly, the servo piston 8 carries out a stroke toward a lower side at a moving amount of the spool 9.

In this case, a seal member 27 is fixed to the servo piston 8 by a snap ring 28, and seals in the inner side of the servo piston in such a manner that the pressure oil in the outer side of the spool 9 does not leak to the external portion.

A description will be given of an operation of the volume control apparatus in accordance with the present

embodiment.

In a steady state, as shown in Fig. 2, the spool 9 is at a standstill under a state in which a force downward in the drawing in correspondence to the pilot pressure applied to the pressure receiving surface 9a and a spring force K (obtained by subtracting the spring force of the spring 10 from the spring force of the spring 11) upward in the drawing applied by the spring 11 and the spring 10 are balanced. Further, the opening areas of the throttle 23 and the throttle 25 are regulated, and the thrust generated in the servo piston 8 and the thrust obtained by adding the thrust of the opposing piston 7 to the thrust of the piston 4 applied via the cam ring 2 are balanced and at a standstill.

In this case, when the pilot pressure supplied to the pilot port 12 is increased, the force in correspondence to the pilot pressure applied to the pressure receiving surface 9a becomes larger than the spring force K applied by the spring 11 and the spring 10, and the spool 9 carries out a stroke toward a lower side, that is, toward the cam ring 2.

When the spool 9 relatively carries out a stroke toward the cam ring 2 with respect to the servo piston 8, the opening area of the throttle 23 is increased, the pressure oil supplied from the original pressure port 13 to the oil chamber 20 via the driving pressure introducing oil passage 22, the throttle 23, the oil passage 9b, the spring chamber of the spring 10 and the oil passage 9d is increased, and the driving pressure is increased. Accordingly, the thrust

generated in the servo piston 8 becomes larger than the thrust obtained by adding the thrust of the opposing piston 7 to the thrust of the piston 4 applied via the cam ring 2, and the servo piston 8 presses the cam ring 2 against the opposing piston 7, and moves toward the opposing piston 7.

Since the spool 9 moves toward the cam ring 2 and the servo piston 8 moves toward the cam ring 2, the spring 10 is expanded while the length of the spring 11 is not changed, so that the spring force K (obtained by subtracting the spring force of the spring 10 from the spring force of the spring 11) upward in the drawing is increased. Accordingly, the movement of the spool 9 is inhibited. Therefore, the opening area of the throttle 23 is reduced, and the increase of the driving pressure supplied to the oil chamber 20 from the original pressure port 13 via the driving pressure introducing oil passage 22, the throttle 23, the oil passage 9b, the spring chamber of the spring 10 and the oil passage 9d is inhibited.

As mentioned above, the spool 9 stands still at a position where the downward force corresponding to the pilot pressure and the spring force K upward in the drawing applied by the spring 10 are balanced. In other words, the spring 10 is positioned at the lower position expanding from the state shown in Fig. 2.

The servo piston 8 stands still at the position corresponding to the position at which the spool 9 is positioned in a standstill state, in a state in which the

thrust generated in the servo piston 8 is balanced with the thrust obtained by adding the thrust of the opposing piston 7 to the thrust of the piston 4 applied via the cam ring 2.

As a result, the cam ring 2 moves further closer to the opposing piston 7 rather than the state in Fig. 2 by the servo piston 8, and the volume of the radial piston pump 1 is reduced.

On the other hand, when the pilot pressure supplied to the pilot port 12 is reduced from the state in Fig. 2, the force in correspondence to the pilot pressure applied to the pressure receiving surface 9a becomes smaller than the spring force K applied by the spring 11 and the spring 10, and the spool 9 carried out a stroke toward an upper side, that is, in a direction of moving apart from the cam ring 2.

When the spool 9 carries out the stroke in the direction of relatively moving apart from the cam ring 2 with respect to the servo piston 8, the opening area of the throttle 25 is increased, the pressure oil discharged from the oil chamber 20 to the tank 26 via the oil passage 9c, the throttle 25 and the tank discharging oil passage 24 is increased, and the driving pressure is reduced. Accordingly, the thrust generated in the servo piston 8 becomes smaller than the thrust obtained by adding the thrust of the opposing piston 7 to the thrust of the piston 4 applied via the cam ring 2, and the servo piston 8 moves in the direction of moving apart from the opposing piston 7 while being pressed by the cam ring 2.

Since the spool 9 moved in the direction of moving apart from the cam ring 2, and the servo piston 8 moves in the direction of moving apart from the opposing piston 7, the spring 10 is contracted while the length of the spring 11 is not changed, so that the upward spring force K applied by the spring 11 and the spring 10 is reduced. Accordingly, the movement of the spool 9 is inhibited. Accordingly, the opening area of the throttle 25 is reduced, the pressure oil discharged from the oil chamber 20 to the tank 26 via the oil passage 9c, the throttle 25 and the tank discharging oil passage 24 is reduced, and the reduction in the driving pressure is inhibited.

As mentioned above, the spool 9 stands still at a position where the downward force in correspondence to the reduced pilot pressure and the upward spring force K applied by the spring 11 and the spring 10 are balanced. In other words, the spool 9 is positioned at a further upper position where the spring 10 is contracted from the state in Fig. 2.

Further, the servo piston 8 stands still at a position corresponding to the position where the spool 9 is positioned at a standstill, in a state in which the thrust generated in the servo piston 8 is balanced with the thrust obtained by adding the thrust of the opposing piston 7 to the thrust of the piston 4 applied via the cam ring 2.

As a result, the cam ring 2 moves toward the side of moving further apart from the opposing piston 7 in comparison with the state in Fig. 2 by the servo piston 8, and the

volume of the radial piston pump 1 is increased.

As mentioned above, in accordance with the present embodiment, the servo piston 8 presses the cam ring 2 following to the spool 9, and positions the cam ring 2 at the position corresponding to the pilot pressure so as to regulate the volume. Accordingly, the same servo mechanism as that in accordance with the prior art can be achieved. Further, since the volume control apparatus has the spool 9 built in the servo piston 8, the place product becomes small, and the weight becomes light. Therefore, the radial piston pump 1 is downsized, the weight is reduced, and the freedom in arrangement is improved.

In this case, when rotating a head portion of the adjusting screw 15 so as to regulate the screwing position with respect to the case 14, it is possible to change the length of the spring regulated by the adjusting screw 15. Accordingly, a corresponding relation between the pilot pressure (the volume control pressure) and the actual volume of the radial piston pump 1 can be set. When the corresponding relation between the pilot pressure and the volume is set to a desired relation in accordance with the adjustment of the adjusting screw 15, the adjusting screw 15 is fixed to the case 14 by the lock nut 16.

Various modifications (applications) can be employed in the present embodiment mentioned above. A description will be given below of the various modifications by attaching the same reference numerals to the same constituting elements as

those in Figs. 1 and 2 and omitting overlapping portions.

Fig. 3 shows a volume control apparatus having a structure corresponding to Fig. 2, and a description will be given of different parts from those in Fig. 2. In Fig. 2, the original pressure port 13 is formed in the lower side of the servo piston 8 close to the cam ring 2, and the pilot port 12 is formed in the upper side apart from the cam ring 2. On the contrary, in Fig. 3, the pilot port 12 is formed in the lower side of the servo piston 8 close to the cam ring 2, and the original pressure port 13 is formed in the upper side apart from the cam ring 2. In correspondence to this structure, the pressure receiving surface 9a of the spool 9 is formed in the lower side closer to the cam ring 2 in Fig. 3 than in Fig. 2. Further, the seal member 27 is provided in the upper side of the servo piston 8 and the spool 9 in Fig. 2, however, is provided in the lower side of the servo piston 8 and the spool 9 in Fig. 3.

In this case, in Figs. 2 and 3, two springs 10 and 11 are attached to the spool 9, and the force corresponding to the pilot pressure is balanced with the spring force, however, one spring may be provided.

Fig. 4 shows a structure of the volume control apparatus in correspondence to Figs. 2 and 3, and shows an embodiment in which only one spring 10 is attached to the spool 9. In other words, the spring 10 expanding and contracting in the same direction as the direction of stroke of the spool 9 is received in the inner side of the spool 9.

One end of the spring 10 is brought into contact with the spool 9, and another end of the spring 10 is brought into contact with the adjusting screw 15.

In a steady state, as shown in Fig. 4, the spool 9 is at a standstill under a state in which a downward force in correspondence to the pilot pressure applied to the pressure receiving surface 9a and an upward spring force K applied by the spring 10 are balanced. Further, the opening areas of the throttle 23 and the throttle 25 are regulated, and the thrust generated in the servo piston 8 and the thrust obtained by adding the thrust of the opposing piston 7 to the thrust of the piston applied via the cam ring 2 are balanced and at a standstill.

In this case, when the pilot pressure supplied to the pilot port 12 is increased, the force in correspondence to the pilot pressure applied to the pressure receiving surface 9a becomes larger than the spring force K applied by the spring 10, and the spool 9 carries out a stroke toward a lower side, that is, toward the cam ring 2.

When the spool 9 relatively carries out a stroke toward the lower side, that is, toward the cam ring 2 with respect to the servo piston 8, the opening area of the throttle 23 is increased, the pressure oil supplied from the original pressure port 13 to the oil chamber 20 via the driving pressure introducing oil passage 22, the throttle 23, the oil passage 9b and the oil passage 9d is increased, and the driving pressure is increased. Accordingly, the thrust

generated in the servo piston 8 becomes larger than the thrust obtained by adding the thrust of the opposing piston 7 to the thrust of the piston 4 applied via the cam ring 2, and the servo piston 8 presses the cam ring 2 against the opposing piston 7, and moves toward the opposing piston 7.

Since the servo piston 8 moves relatively downward, that is, toward the cam ring 2, with respect to the spool 9, whereby the opening area of the throttle 23 is reduced, and the increase of the driving pressure supplied to the oil chamber 20 from the original pressure port 13 via the driving pressure introducing oil passage 22, the throttle 23, the oil passage 9b and the oil passage 9d is inhibited.

As mentioned above, the spool 9 stands still at a position where the downward force corresponding to the pilot pressure and the upward spring force K applied by the spring 10 are balanced. In other words, the spring 10 is positioned at the lower position contracting from the state shown in Fig. 4.

The servo piston 8 stands still at the position corresponding to the position at which the spool 9 is positioned in a standstill state, in a state in which the thrust generated in the servo piston 8 is balanced with the thrust obtained by adding the thrust of the opposing piston 7 to the thrust of the piston 4 applied via the cam ring 2.

As a result, the cam ring 2 moves further closer to the opposing piston 7 rather than the state in Fig. 4 by the servo piston 8, and the volume of the radial piston pump 1 is

reduced.

On the other hand, when the pilot pressure supplied to the pilot port 12 is reduced from the state in Fig. 4, the force in correspondence to the pilot pressure applied to the pressure receiving surface 9a becomes smaller than the spring force K applied by the spring 10, and the spool 9 carried out a stroke toward an upper side, that is, in a direction of moving apart from the cam ring 2.

When the spool 9 carries out the stroke relatively toward the upper side, that is, in the direction of relatively moving apart from the cam ring 2 with respect to the servo piston 8, the opening area of the throttle 25 is increased, the pressure oil discharged from the oil chamber 20 to the tank 26 via the oil passage 9c, the throttle 25 and the tank discharging oil passage 24 is increased, and the driving pressure is reduced. Accordingly, the thrust generated in the servo piston 8 becomes smaller than the thrust obtained by adding the thrust of the opposing piston 7 to the thrust of the piston 4 applied via the cam ring 2, and the servo piston 8 moves in the direction of moving apart from the opposing piston 7 while being pressed by the cam ring 2.

Since the servo piston 8 moves relatively upward, that is, in the direction of moving apart from the cam ring 2, with respect to the spool 9, the opening area of the throttle 25 is reduced, the pressure oil discharged from the oil chamber 20 to the tank 26 via the oil passage 9c, the

throttle 25 and the tank discharging oil passage 24 is reduced, and the reduction in the driving pressure is inhibited.

As mentioned above, the spool 9 stands still at a position where the downward force in correspondence to the reduced pilot pressure and the upward spring force K applied by the spring 10 are balanced. In other words, the spool 9 is positioned at a further upper position where the spring 10 is expanded from the state in Fig. 4.

The servo piston 8 stands still at a position corresponding to the position where the spool 9 is positioned at a standstill, in a state in which the thrust generated in the servo piston 8 is balanced with the thrust obtained by adding the thrust of the opposing piston 7 to the thrust of the piston 4 applied via the cam ring 2.

As a result, the cam ring 2 moves toward the side of moving further apart from the opposing piston 7 in comparison with the state in Fig. 4 by the servo piston 8, and the volume of the radial piston pump 1 is increased.

Either the embodiments mentioned above are based on the one-flow-way type radial piston pump in which the discharging direction is fixed. However, the present invention can be applied to an alternating type radial piston pump in which the discharging direction can be changed to two directions.

Fig. 5A is a view corresponding to Fig. 1. The case 14 is provided with the servo piston 8 having the spool 9 built-in in the same manner as the structure shown in Fig. 2, and

the opposing piston 7, in such a manner as to oppose to each other so as to hold the rotation axis therebetween. In the same manner, the servo piston 18 having the spool 19 built-in and the opposing piston 17 are provided in such a manner as to oppose to each other so as to hold the rotation axis therebetween. Further, the servo piston 8 having the spool 9 built-in, and the servo piston 18 having the spool 19 built-in are provided at opposing positions with holding the cam ring 2 therebetween. Accordingly, it is possible to make the cam ring 2 to be eccentric in both sides with respect to the center of the rotation axis, and the center of the piston valve 5.

The radial piston pump 1 capable of flowing in two direction shown in Fig. 5A is used as a constituting element of a hydraulic circuit shown in Fig. 5B. The hydraulic circuit in Fig. 5B is used in a hydro static transmission (HST) vehicle such as a bulldozer or the like. In the HST vehicle, right and left traveling bodies (wheels or crawler belts) of a vehicle body are independently driven by the HST provided respectively in right and left sides. In a hydraulic circuit of the HST, in the case that the pressure oil is discharged from one discharge port of a hydraulic pump 61 and the pressure oil is flowed into one port of a hydraulic motor 60, the hydraulic motor 60 rotates forward and the vehicle moves forward. Further, in the case that the pressure oil is discharged from another discharge port of the hydraulic pump 61 and the pressure oil is flowed into another

port of the hydraulic motor 60, the hydraulic motor 60 rotates backward and the vehicle moves backward. A gear change is performed by changing the volume of the hydraulic pump 61 and the hydraulic motor 60.

As shown in Fig. 5A, in order to change the moving direction of the vehicle to the forward moving direction and the backward moving direction, and change the volume of the radial piston pump 1 in each of the directions, there are provided a switching valve 40 for changing between the forward movement and the backward movement, an electromagnetic proportional control valve 31 for controlling the volume at a time of forward moving, and an electromagnetic proportional control valve 32 for controlling the volume at a time of moving backward. As hydraulic pressure sources of the pilot pressure and the driving pressure supplied to the servo pistons 8 and 18, there are provided respectively pilot hydraulic pressure sources 27 and 27 and a driving pressure source 29. In this case, the driving pressure source 29 can employ the radial piston pump 1 itself.

When a forward movement command signal S1 is applied to an electromagnetic solenoid of the electromagnetic proportional control valve 31 by an operation lever (not shown) or the like, the electromagnetic proportional control valve 31 is driven to an open side, and the pilot pressure is applied to a pilot port 40d of the switching valve 40 via the electromagnetic proportional control valve 31 from the pilot

hydraulic pressure source 27. Accordingly, the switching valve 40 is changed to a forward moving position 40a from a neutral position 40c. Therefore, the electromagnetic proportional control valve 31 is opened at an opening degree in proportional to the forward movement command signal S1, and the pilot pressure of the pilot hydraulic pressure source 27 is reduced to the pilot pressure in correspondence to the opening degree of the electromagnetic proportional control valve 31, and is supplied to the pilot port 12 of the servo piston 8 via the switching valve 40 and the oil passage 41.

The driving pressure of the driving pressure source 29 is supplied to the original pressure port 13 of the servo piston 8 via the switching valve 40 and the oil passage 42. Further, the driving pressure of the driving pressure source 29 is supplied to the oil chamber 28 of the opposing piston 7 opposing to the servo piston 8 via the switching valve 40 and the oil passage 46. In this case, the opposing piston 17, the pilot port 12 of another servo piston 18 and the original pressure port 13 are respectively communicated with the tank 26 via an oil passage 43, an oil passage 44, an oil passage 45 and the switching valve 40.

Accordingly, the servo piston 8 and the spool 9 are operated as described in Fig. 2, and the cam ring 2 is made eccentric to the position corresponding to the pilot pressure supplied to the pilot port 12 of the servo piston 8. The cam ring 2 is made eccentric to a left side in the drawing with respect to the rotation axis. Accordingly, the pressure oil

at a volume corresponding to the forward movement command signal S1 is discharged from one discharging direction of the radial piston pump 1, and the vehicle travels forward at a speed corresponding to the forward movement command signal S1.

On the contrary, in the case that a backward movement command signal S2 is applied to an electromagnetic solenoid of the electromagnetic proportional control valve 32 by the operation lever or the like, the electromagnetic proportional control valve 32 is driven to an open side, and the pilot pressure is applied to a pilot port 40e of the switching valve 40 from the pilot hydraulic pressure source 27 via the electromagnetic proportional control valve 32. Accordingly, the switching valve 40 is changed to a backward moving position 40b from the neutral position 40c. Therefore, the electromagnetic proportional control valve 32 is open at an opening degree in proportion to the backward movement command signal S2, and the pilot pressure of the pilot hydraulic pressure source 27 is reduced to a pilot pressure corresponding to the opening degree of the electromagnetic proportional control valve 32 and is supplied to the pilot port 12 of the servo piston 18 via the switching valve 40 and the oil passage 44.

The driving pressure of the driving pressure source 29 is supplied to the original pressure port 13 of the servo piston 18 via the switching valve 40 and the oil passage 45. Further, the driving pressure of the driving pressure source 29 is supplied to an oil chamber 38 of the opposing piston 17

opposing to the servo piston 18 via the switching valve 40 and the oil passage 43. In this case, the opposing piston 7, the pilot port 12 of another servo piston 8 and the original pressure port 13 are communicated respectively with the tank 26 via the oil passage 46, the oil passage 41, the oil passage 42 and the switching valve 40.

Accordingly, the servo piston 18 and the spool 19 are operated as described in Fig. 2, and the cam ring 2 is made eccentric to a position corresponding to the pilot pressure supplied to the pilot port 12 of the servo piston 18. The cam ring 2 is made eccentric to a right side in the drawing with respect to the rotation axis. Therefore, the pressure oil at the volume corresponding to the backward movement command signal S2 is discharged from another discharging direction of the radial piston pump 1, and the vehicle travels backward at a speed corresponding to the backward movement command signal S2.

In this case, the structure in Fig. 5A is based on the volume control apparatus having the structure shown in Fig. 2, however, can employ the volume control apparatus having the structure shown in Fig. 3 or 4.

The embodiment mentioned above is described on the basis of the radial piston pump, however, can be applied also to the radial piston motor as it is.

Further, the embodiments mentioned above are not limited to the volume control apparatus for the radial piston pump or motor, but can be applied to a volume control

apparatus for a swash plate type axial piston pump or motor.

Fig. 6 shows an embodiment employing the volume control apparatus shown in Fig. 2 as a volume control apparatus for regulating the volume by pressing a swash plate 50 of an axial piston pump 55 and oscillating the swash plate 50.

The servo piston 8 having the spool 9 built-in in the same manner as the structure shown in Fig. 2 is brought into contact with the swash plate 50, and the servo piston 8 presses the swash plate 50 on the basis of a thrust corresponding to the pilot pressure so as to oscillate the swash plate 50, and positions the swash plate 50 at a position tilted from a rotation axis 51. A stroke amount of a piston 54 is determined in correspondence to a tilting amount of the swash plate 50 with respect to the rotation axis 51, and a volume of the axial piston pump 55 is determined. In Fig. 6, the structure is made such that a position of a supporting point of a ball for supporting the tilting motion of the swash plate 50 is displaced to a left side in the drawing from a position of a working point of a resultant force applied to the swash plate 50 by the shoe of the piston 54, thereby generating a force opposing to the thrust of the servo piston 8 existing in a right side in the drawing with respect to the ball, and achieving the same function as that of the opposing piston 7 in Fig. 1.

In this case, the structure in Fig. 6 can be applied also to the axial piston motor as it is.

Further, the structure in Fig. 6 is based on the volume

control apparatus having the structure shown in Fig. 2, however, may employ the volume control apparatus having the structure shown in Fig. 3 or 4.

Further, the apparatus having the structure shown in Figs. 2, 3 and 4 is not limited to the apparatus for positioning the cam ring or the swash plate of the hydraulic pump or motor, and may be used as a positioning apparatus for positioning the other members to be positioned than the cam ring and the swash plate to a desired position.